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Modeling and analysis of a robust thermal control system based on forced convection thermal switches

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ABSTRACT

There is a critical need, not just in the Department of Defense (DOD) but the entire space industry, to reduce the development time and overall cost of satellite missions. To that end, the DOD is actively pursuing the capability to reduce the deployment time of a new system from years to weeks or even days. The goal is to provide the advantages space affords not just to the strategic planner but also to the battlefield commanders. One of the most challenging aspects of this problem is the satellite's thermal control system (TCS). Traditionally the TCS must be vigorously designed, analyzed, tested, and optimized from the ground up for every satellite mission. This "reinvention of the wheel" is costly and time intensive. The next generation satellite TCS must be modular and scalable in order to cover a wide range of applications, orbits, and mission requirements. To meet these requirements a robust thermal control system utilizing forced convection thermal switches was investigated. The problem was investigated in two separate stages. The first focused on the overall design of the bus. The second stage focused on the overarching bus architecture and the design impacts of employing a thermal switch based TCS design. For the hot case, the fan provided additional cooling to increase the heat transfer rate of the subsystem. During the cold case, the result was a significant reduction in survival heater power.

Keywords : satellite, thermal control, forced convection, thermal switch, responsive space

1. INTRODUCTION

There has been a growing need in the Department of Defense (DOD) to make space more responsive and cost effective. Instead of taking years to design and deploy a new satellite, the goal is weeks or even days. The goal is to extend the advantages space affords from the strategic planner to the battlefield commanders. The ability to launch a new space asset within days or hours of a battlefield commander's request will maintain the asymmetric advantage in future conflicts. For space to become operationally responsive, satellites must be easily manufactured, assembled, tested, and prepared for launch in a military depot style environment. Large geosynchronous satellites will continue to play an important role in space activities, but to achieve the goals of responsive space, components and systems will have to be standardized and simple, which translates to an increasing usage of small satellites.

To meet this challenge, the methodologies used to design, manufacture, test, launch, and deploy satellites must radically change. One of the most challenging aspects of this problem is the satellite's Thermal Control Subsystem (TCS). Traditionally, the TCS is vigorously designed, analyzed, and optimized for every satellite mission. This "reinvention of the wheel" is costly and time intensive. The next generation satellite TCS must be robust, modular, and scalable to meet the needs of a wide range of missions, payloads, and thermal requirements. In order to meet these needs, a Forced Air Convection Thermal Switch (FACTS) concept was investigated. The modeling and analysis of such a TCS was the focus of this paper.

2. THERMAL CONTROL APPROACH

The FACTS concept is similar to terrestrial systems used to cool electronics, where a DC axial fan forces air through the system to cool components. The primary difference is this system is sealed and forced convection within the enclosure transfers heat from the components to the walls of the enclosure where it is conducted and radiated to the

exterior of the bus. This concept is not completely new. The first satellites used air convection for cooling, and this approach is still used to cool electronics at high altitudes¹.

There are significant advantages and disadvantages to this type of system for satellites. Forced convection provides higher heat transfer coefficients than conduction. Therefore, a more efficient design is possible, and the thermal gradient over the subsystem can be reduced. Another advantage is that a simple DC axial fan acts as a thermal switch. When heat loads are high, the fan is switched on and provides additional cooling through convection. When loads are reduced, the fan is turned off reducing heat transfer. Finally, sealing the enclosure provides significant advantages for depot-style operations. It reduces the clean room requirements for the depot. Also, thermal joint quality requirements are reduced since the components are sealed in an atmospheric environment. Of course, there are significant challenges that must be addressed.

The biggest disadvantage to a forced convection system is the added mass required to maintain an internal pressure of 1 atm in the hard vacuum of space. Sealing the box to prevent leakage and eventual failure is also a critical design factor. Finally, adding a fan increases both the power requirements of the bus and the complexity of the attitude control system. A standard DC axial fan capable of producing a flow rate of 30 CFM against a pressure of 6 mmHg₂O requires approximately 4 W of power. This adds stress to the power system and an added load on the TCS. The complexity added to the attitude control subsystem is the addition of a rotating component with its own vibration spectrum that turns off and on almost instantaneously. However, the advantage of a modular, robust system outweighs the disadvantages when a short turn-around-time becomes more important than mass.

2.1 Satellite Bus Summary

The traditional approach to satellite design is a customized and highly optimized satellite bus. The primary design driver is to minimize mass but often at the expense of time and money. To meet the goals of Operationally Responsive Space (ORS), the satellite must be adaptable to different missions, changing threats, and emerging technologies. One philosophical approach to achieve these goals in the near term is to separate the design and engineering of the payload from the bus. The bus would have a standard design and that provides a specific set of baseline capabilities. If additional capability is required by the payload, then the capability would have to be provided by the payload. The payload would then be integrated to the bus through standard mechanical, electrical, thermal, and software interfaces. It should be noted that, according to this philosophy, there will be some payloads that can not be economically accommodated by the standard bus, and a unique system will have to be designed. This philosophy is not new and has analogies in the computer and automotive industry. A supplier, such as Dell, has a standard model that will meet the needs of the majority of the market. For those users that need additional features, such as a faster processor or better graphics the appropriate upgrades are made to the standard model before the unit is shipped to the customer. For the user that requires a top of the line system, often times a custom built system provides the only economic solution.

Currently, there are many companies that have developed or are developing low-cost, modular, satellite bus architectures that are specifically directed toward ORS. The disadvantage with most standardized bus development programs is that the bus eventually becomes obsolete and must be completely redesigned as new technologies are developed. One of the goals of the ORS program is the development of technologies that provide robust and flexible bus designs. For example, a space plug-and-play concept similar to PC based plug-and-play USB connectivity is being investigated for the avionics system². Plug-and-play addresses the software and electrical interfaces, but other efforts are needed to address the mechanical and thermal interfaces.

Regardless of the design philosophy, a certain level of fidelity of the bus design is needed before the basic requirements for the thermal control subsystem can be identified. Because of launch vehicle limitations, ORS missions will likely be relegated to 450 kg class satellites. Using this basic assumption, the capabilities that a small satellite bus can provide can be determined. In a previous effort, two satellite busses were sized to meet responsive space needs³. The first bus provided minimal capabilities, while the other provided significantly improved capabilities. These two busses represent a lower and upper bounds for design. The focus of this paper is the TCS design for the lower capability bus, which is summarized below.

Table 1: Satellite subsystem summary

Subsystem	Capability	Mass	Power	Size
		[kg]	[W]	[cm]
Attitude Determination & Control (ADC)	1°-5° attitude control	10.3	18.5	30 x 24 x 12
Telemetry, Tracking, & Command (TTC)	1 Mbs, S-band transmitter	2.8	7.4	9.8 x 9.6 x 7.2
Navigation & Guidance (NG)	12 channel GPS receiver	0.02	0.8	7.0 x 4.5 x 1.0
Command & Data Handling (CDH)	Plug-n-play USB architecture	15.2	50	34 x 25 x 20
Power Management (PM)	500 W, 3J array, PPT system	18.3	70.3	25 x 23 x 21
Structure	Al Honeycomb Panels	21.5	n/a	27 x 40.5 x 71
Propulsion	No propulsion system	0	0	0 x 0 x 0
		68.1	147.0	27 x 40.5 x 71

In addition to the above, the location and orientation of the components must be determined. To simplify the integration of the components into the bus, they were separated by subsystem and sealed in enclosures. This provides to advantages. The first is storage in a depot-style environment similar to airplane operations. Initially, for ORS to be feasible, operations will have to be conducted in a depot-style environment where mission orders are received and components are quickly integrated into the bus and tested. Integrating the subsystems into enclosures protects the components and can reduce the requirements on the storage facility if the enclosures are hermetically sealed. The second advantage enclosing the subsystem provides is simplifying interface standards. Since thermal design is separated into two parts, overall bus design and component specific design, a natural breakpoint occurs at the interface between the bus and the subsystems. Rather than having to specify interface standards for every type of component, standards would only have to be created for the subsystem enclosure/bus interface. By separating at that location, the subsystem supplier would be responsible for developing the thermal design of the components inside the enclosure; whereas, the system integrator would be responsible for developing the overall thermal control of the bus. The interface between the bus and the subsystems enclosures would be dictated by a thermal design standard that both parties would have to follow.

The figure below shows the location and orientation of the subsystem enclosures for the bus. In addition, the figures show the face that is reserved as the interface plane between the bus and the payload. At this location there is no heat transfer between the bus and the payload or between the bus and the external environment

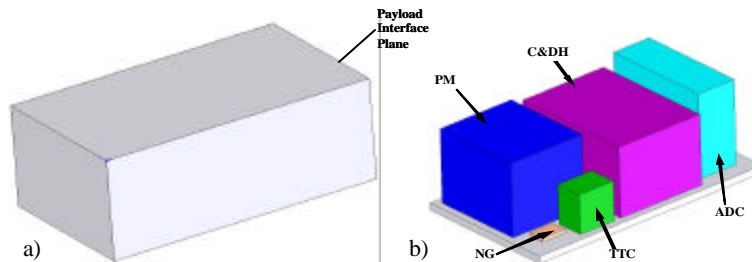


Figure 1: Solid models of the bus a) exterior and b) subsystem location and orientation

2.2 Energy Balance

Essentially, the primary task of the thermal engineer is to balance the thermal energy of the satellite to ensure all of the internal components remain within their acceptable temperature limits during the worst hot and cold cases. External and internal heat generation must be properly balanced with the excess heat radiated to space. Figure 2 summarizes the sources and sinks for a satellite in low earth orbit. A simple energy balance analysis between the satellite and the space environment can be used to determine whether or not the satellite has enough surface area to maintain its temperature within acceptable limits for the hot case. In addition, it can be used to size survival heater power to maintain the temperature within acceptable limits for the cold case.

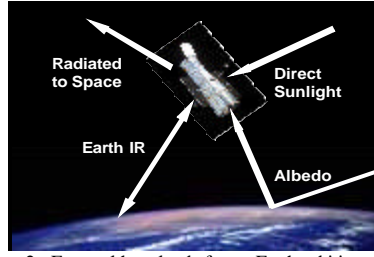


Figure 2: External heat loads for an Earth orbiting satellite

The actual temperature of space is 4 K; however, as a first order approximation the temperature of space can be assumed to be 0 K. Substituting in expressions for the heat loads, the energy balance equation is⁴:

$$\epsilon A_{rad} T_s^4 = \epsilon A_s F_{s,e} I_{EIR} + \alpha A_{\perp} I_{sun} + \alpha a A_s F_{s,se} I_{sun} + Q_{Internal} \quad (1)$$

where ϵ is the emissivity of the spacecraft, σ is the Stefan-Boltzmann constant [$\text{W/m}^2\text{-K}^4$], A_{rad} is the radiator surface area [m], T_s is the average temperature of the spacecraft [K], A_s is the surface area [m], $F_{s,e}$ is the view factor between the spacecraft and the Earth, I_{EIR} is the intensity of the Earth IR, α is the surface solar absorptivity, A_{\perp} is the area perpendicular to the sun [m], and I_{sun} is the solar heat flux [W/m^2], a is the Earth albedo coefficient, $F_{s,se}$ is the view factor between the spacecraft and the sunlit Earth, and $Q_{Internal}$ is the internal heat generation [W]. This equation provides a first order approximation of the radiator area need for the hot case and the heater power needed for the cold case.

The worst hot and cold cases are dependent on the orbit. Since specific orbits are largely unknown for ORS, the TCS must be adaptable to all low earth orbits. However, the number of possible orbits was limited to circular orbits for simplicity. Using these constraints, the worst hot case orbit has a beta angle of 90°, and the worst cold case orbit has a beta angle of 0°. The worst case orientations are shown on Figure 3, and the values associated with these two cases are summarized on Table 2.

For the first order approximation of the energy balance, the internal heat load for the hot case, which was summarized on Table 1, is 147.0 W. As for the cold case, the internal heat load was assumed to be 50 W. This value represents the minimal power needed to keep the satellite operational while in safe mode. Next, it was assumed that the surface was painted white, and only five surfaces were available for radiation to space. The remaining surface was reserved as the interface surface for the payload. An emissivity of 0.88 and an absorptivity of 0.22 were used for the white paint. Finally, the temperature limits for the satellite were dependent on the components. As a first order estimate, the upper temperature limit was constrained to 303 K, and the lower limit was constrained to 273 K. These values represent the components with the tightest temperature constraints, which were the momentum wheels and the Li-ion batteries.

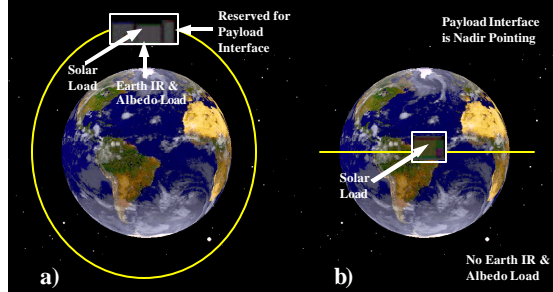


Figure 3: Worst case orientations a) hot case and b) cold case

Table 2: Worst hot and cold case conditions⁵

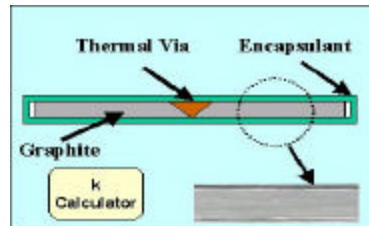
	Hot Case	Cold Case
Eclipse Percent	0	0.43
Solar Constant [W/m^2]	1414	1322
Albedo Coefficient	0.57	0.18
Earth IR [W/m^2]	275	218
Internal Heat [W]	147	50
Temperature Limit [K]	303	273
Area - to Sun [m^2]	0.1917	0.1094
Area - to Earth [m^2]	0.2876	0

Using the energy balance equation and the parameters above, the radiator area required to keep the bus below 303 K was 0.76 m^2 and the resulting cold case temperature was 204.6 K. The total available radiator area of the bus was 1.07 m^2 . If the surface area was increased to the total available radiator area, the hot case temperature was reduced to 278.3 K, and the cold case temperature was reduced to 187.9 K. To increase the cold case temperature to acceptable levels, a survival heater power of 240 W would be required. A passive thermal control system incorporating survival heaters would satisfy the thermal requirements. However, an active system might be needed because of the large survival heater power requirement.

3. THERMAL MODELING AND ANALYSIS

Because the subsystems were housed in separate enclosures, the design of the TCS was split into two parts. The first part, which focused on the design of the overall bus TCS, emphasized the conduction of heat from the subsystems through the bus structure to the exterior of the satellite where it can be radiated to space. Initially, the subsystems were modeled as simple aluminum enclosures with uniform heat loads. After the heat conduction path through the bus was designed, the focus turned to the subsystems, which was the focus of the second part of the design. Finally, the bus model and the subsystem models were integrated, and the final design was analyzed.

Initially, two design options were considered for the bus. The first option considered was a bus structure constructed of bare aluminum honeycomb panels. The second design incorporated a honeycomb electronics shelf with an APG core to improve the lateral conductivity of the panel. The design was based on k-Technologies' k-Core concept. A schematic of the k-Core concept is shown in below.

Figure 4: k-Technologies patented k-Core material system⁶

3.1 Bus Thermal Model

The bus structure and the subsystem enclosures were modeled Thermal Desktop (TD). The subsystems were modeled as Al-2024 enclosures with wall thicknesses of 1.5875 mm and a thermal conductivity of 185 W/m-K. The edges between the different sides of the enclosures were assumed to be in perfect contact, which is the equivalent of a continuous material around the corners. The conductivity of the interface between each of the subsystem enclosures and

the shelf was controlled using surface contact conductors. Conservatively, a joint conductivity of 110 W/m-K was assumed for bare aluminum interfaces⁷. As for the electronics shelf, it was modeled as an aluminum honeycomb panel with a 1 mm APG core encapsulated in the face sheets. As a result, the face sheets for the electronics shelf were 2.6 times thicker than the face sheets used for the other panels. The interface conductivity between the face sheets and the core was controlled using surface contact conductors. The other panels were modeled as two face sheets with a contactor to control the conductivity of the honeycomb core. Initially, a honeycomb core transverse conductance of 250 W/m²-K was used. The panels were also assumed to be in perfect contact with one another.

As for the boundary conditions, the internal heat loads for each subsystem were evenly distributed over all six surfaces of the enclosure. The external loads were applied using surface heat loads. The solar loads for the hot and cold cases were 312 W/m² and 166 W/m², respectively. The combined Earth IR and albedo load for the hot case was 414 W/m². White paint with an ϵ of 0.88 and an α of 0.22 was used for the exterior of the satellite. RadCAD was used to calculate the radiation exchange factors with space. Radiation within the bus was included in the calculations. The interior surfaces were painted black to enhance radiative heat transfer.

The results from TD are presented below in Figure 5 and 6. Both the satellite exterior and subsystem temperature distributions are shown. For the hot case, the maximum temperatures for the ADC, NG, and TTC subsystems were 305.3, 294, and 305.9 K at the top surface of the enclosure. More importantly, the enclosure interface temperatures for the three subsystems were less than 300 K. With a proper subsystem design, the maximum component temperature can be reduced to meet the thermal requirements for each of the individual components. Conversely, the CDH and PM subsystems exceeded the upper limit by 18 K and nearly 38 K, respectively, and the enclosure interface temperature for the PM subsystem exceeded the limit. To reduce the interface temperatures for the CDH and PM subsystems, RTV was added to the interface. RTV improves the interface conductivity; a conservative value of 435 W/m²-K was assumed⁶. As a result, the enclosure interface temperatures for all of the subsystems were below 303 K. The maximum temperatures were still above the limit, but this was addressed during subsystem detailed design. The effect on the cold case temperatures was minimal.

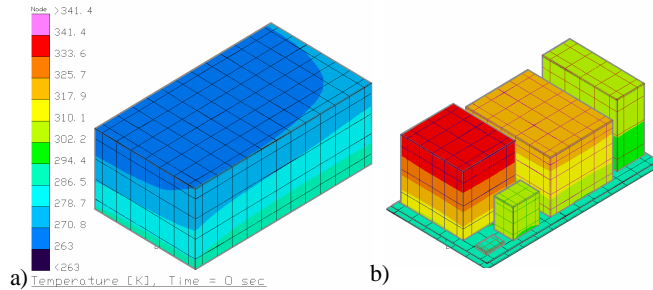


Figure 5: Hot case results for the model a) the temperature distribution over the exterior of the bus and b) the temperature distribution over the subsystem enclosures

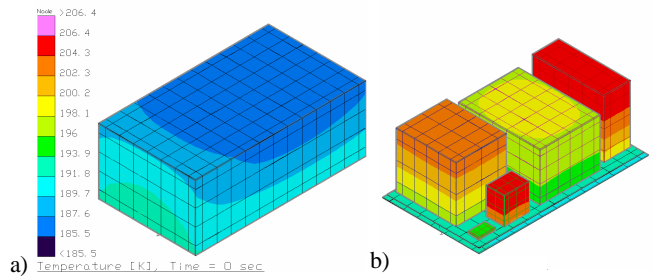


Figure 6: Cold case results for the model a) the temperature distribution over the exterior of the bus and b) the temperature distribution over the subsystem enclosures

3.2 Command and Data Handling Subsystem Detailed Model

The focus will turn to the thermal control of the individual subsystems. Two design approaches were considered for each subsystem. The first was a completely passive conduction-based approach. The second was an active approach using forced convection. From the analysis of the detailed bus model, the passive conduction-based scheme will be adequate for the ADC, NG, and TTC subsystems. The enclosure temperatures were well within the component temperature limits for the hot case scenario. Conversely, the CDH and PM subsystems exceeded the limit. For these two systems, the forced air convection cooled option was investigated. However, only the CDH subsystem design will be presented in detail.

The CDH subsystem consisted of four Printed Circuit Boards (PCBs) mounted to a backplane PCB. The boards were modeled as 0.3 cm thick PCBs fabricated out of FR4 2 oz copper. An edge contact conductivity of 17.7 W/m-K was used for the connection between the PCBs and the mounting rails. All of the boards were mounted on 0.5 cm thick aluminum rails to conduct heat to the walls of the enclosure. Finally, with the exception of the processor, the heat loads were applied as uniform loads over the board. The load on the back plane and legacy interface boards was 5 W. It was 10 W for the power management and the processor board. In addition, a processor heat load of 10 W was applied to a 2 cm by 2 cm area on the processor board. The location and orientation of the PCBs are shown on Figure 7.

The passive conduction-based scheme was investigated first. The results are presented in Figure 7. The maximum temperature on the processor was 322.7 K. The temperature over the PCBs ranged between 313 and 318 K. Even if the contact conductivity for all of the components was increased to 10,000 W/m-K, the temperature would still be above the upper temperature limit. Since the system did not meet the temperature requirements, the active system using forced convection was considered.

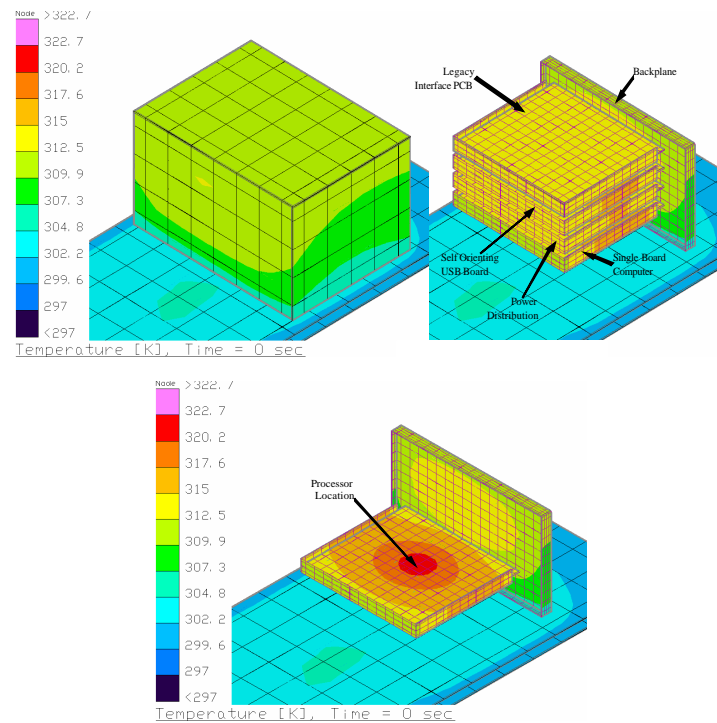


Figure 7: Temperature distribution for the passively cooled CDH subsystem

To investigate the forced convection option, a flow network was added to the previous model. The fan was mounted to the PCBs at location 2, which is shown in Figure 8 with the fan outlet flow path between the boards. The return path is between the processor board and the bottom of the enclosure. For the flow paths between the boards, the spacing between the aluminum side rails was used to calculate the flow area and hydraulic perimeter. For paths 2-3, 3-4, 4-5 and 6-1 shown in Figure 8, the distance between the PCBs and the enclosure was used. Finally, a fan heat load at a ratio of 0.1333 W/CFM was added to the system to account for the additional power requirement of the fan. The ratio was determined from the average power requirements for a DC axial fan⁸.

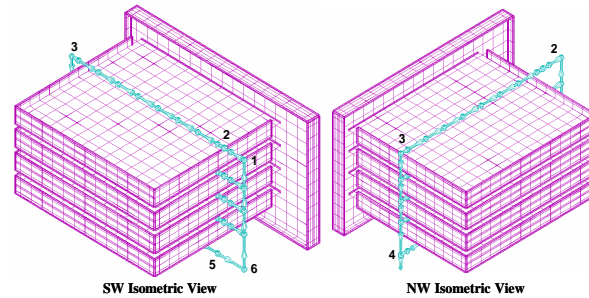


Figure 8: Flow network for the CDH subsystem

The analysis of the system focused on determining the temperature distribution within the enclosure for various flow rates. A parametric analysis was performed to determine the flow rate required to achieve sufficient cooling throughout the enclosure. During the analysis, the flow losses through the system were ignored. Once the flow rate is known, the design reduces to a fan sizing exercise. To ensure that the flow rate did not exceed the capability of DC axial fans, the maximum flow rate was limited to 40 CFM. The maximum and minimum temperatures for the CDH subsystem were plotted, and the results are presented in Figure 9.

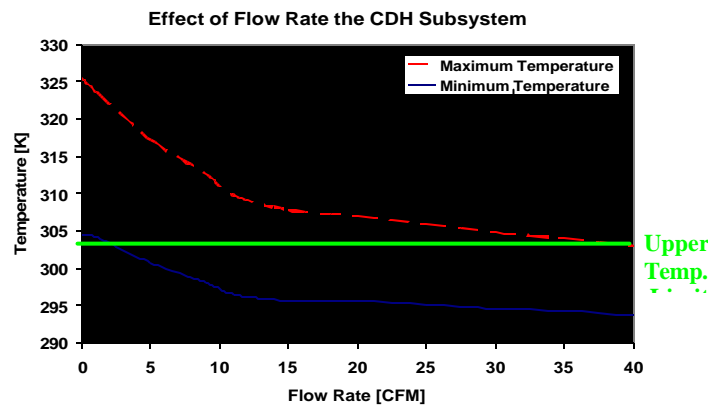


Figure 9: The effect that the fan flow rate has on the CDH subsystem
(The green line represents the upper temperature limit for the components.)

The figure shows that increasing the flow rate significantly improves heat transfer initially. However, as the flow rate increases, the power input for the fan increases, and the return on performance decreases. There are two other points of interest. First, the maximum temperature at 0 CFM is 325.4 K, which is 2.7 K greater than the maximum temperature for the previous conduction case. The reason for this discrepancy is that for the forced convection system the contact conductivity of the aluminum rails was reduced from 10,000 W/m²-K to 110 W/m²-K to be consistent with

the other subsystem. Second, the forced convection system significantly reduces the thermal gradient over the enclosure. When the fan is turned off, the temperature differential is 20.9 K. At 40 CFM, the differential decreases to 9.3K.

At 40 CFM, the maximum temperature is slightly above than the upper temperature limit, which occurs at the location of the processor. The maximum temperature for the system was 303.3 K. The average temperature of the other PCBs is 298 K. Since the design only exceeds the limit by 0.3K, it will probably be acceptable. The temperature distribution throughout the enclosure for a flow rate of 40 CFM is shown in Figure 10.

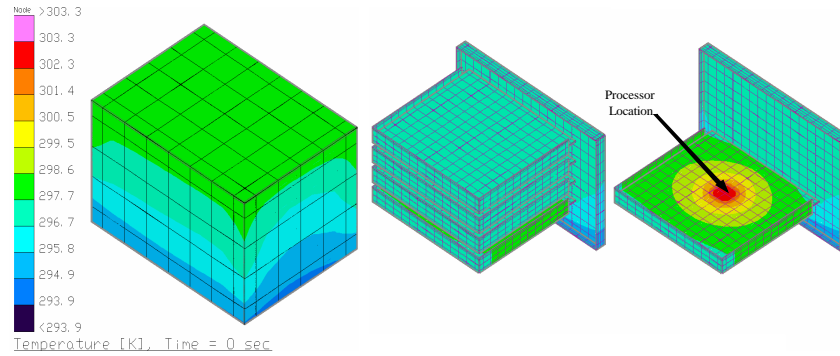


Figure 10: Temperature distribution through the CDH subsystem for a flow rate of 40 CFM

4. REVIEW OF THE FINAL DESIGN

After the detailed bus and subsystem designs were completed, the models were integrated, and the final design was re-evaluated. The temperatures for all of the components were within baseline operating temperature limit of 303 K with the exception of the TTC S-band transmitter. The maximum temperature for the component at the top of the transmitter was 308.8 K. This exceeds the baseline upper temperature limit, which was used as a first approximation. The maximum operating temperature for the transmitter was 333 K, so the transmitter was well within its limits. Next, the maximum temperature for the ADC subsystem was only 295.1 K. To improve the cold case results and to reduce the survival heater requirement, the component temperatures should be as close to the upper limit as possible for the worst case hot condition. By reducing the contact conductivity of the enclosure to $10 \text{ W/m}^2\text{-K}$, the maximum temperature increases to just over 300 K. This can be easily accomplished by reducing the bolt pressure, the bolt spacing, or adding a low conductivity interstitial material, such as felt. Finally, the NG subsystem was also well below the upper temperature limit, which was expected because of its low heat dissipation. The subsystem was moved and mounted to the top of the TTC subsystem enclosure. As a result, the temperature of the component was increased to 303.5 K, which is also well below its maximum operating temperature.

Once the detailed designs for all of the subsystems were integrated into the final design, the designs were re-evaluated and optimized. For the most part, the design remained unchanged. The only exception was in the design of the CDH subsystem where the fan flow rate was reduced to 30 CFM. Even with the reduced flow rate, the temperature at the processor was reduced from 303.3 K to 302.5 K. The reason for the difference between the two was the added fidelity of the model, especially the PM subsystem enclosure. In the final integrated model, the maximum temperature of the PM subsystem was 304.8 K. In the CDH subsystem model, the maximum temperature for the PM subsystem was 325 K because the same aluminum enclosures used in the detailed bus model were used for the other subsystems. By reducing the maximum temperature of the PM subsystem in the final integrated model, the radiation heat transfer between the PM and CDH enclosures was reduced, and the overall temperature of the CDH subsystem was reduced. The final results for the hot case condition are presented in Figure 11.

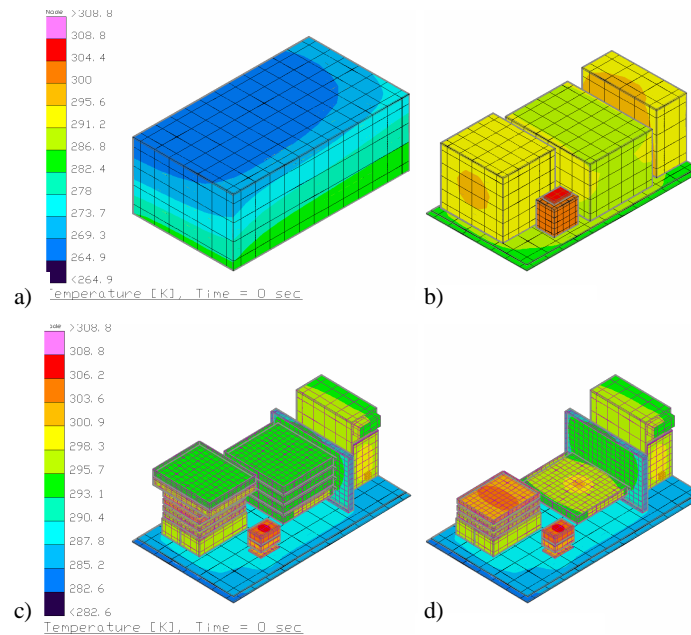


Figure 11: Results for the worst case hot scenario temperature distribution over a) the exterior of the bus, b) subsystem enclosures, c) the components, and d) the hot spot location for each subsystem

As for the cold case, the temperatures of all of the components were well below the minimum temperature limit. To maintain the bus within the baseline temperature limit, an additional heater power of 150 W was needed, which was higher than the total power consumption for the hot case. The difference was caused by the drastic change in the external load between the hot and cold cases. It is important to note that it is also impossible for the worst hot and cold cases to exist for the same orbit. For a more realistic analysis, the worst hot and cold cases were separated by orbit and are outlined below. For each different orbit, the surface properties were tailored and then the heaters were sized.

- 1a. Worst Case for Hot Orbit: Same as before; results are unchanged
- 1b. Cold Case for Hot Orbit: Beta angle of 90° , minimum power output, and an orientation with the payload facing the Earth and the smallest adjacent side receiving the solar load
- 2a. Hot Case for Cold Orbit: Beta angle of 0° , maximum power, and an orientation with largest panels exposed to the solar, Earth IR, and albedo loads
- 2b. Worst Case for Cold Orbit: Same as before

For case number one, the satellite exterior was painted white, and the survival heater power required was reduced to 30 W. For case number two, the exterior was painted green, which increased the solar absorptivity to 0.57. The emissivity was unchanged. The hot case temperatures for the cold orbit (case 2a) were similar to those shown in Figure 11. The survival heater power needed to maintain the minimum temperature was reduced to 40 W.

5. CONCLUSIONS

In the FACTS system concept, the fan provided two distinct functions. The first function was to increase the heat transfer rate through the subsystem to keep the component temperatures below their maximum operating temperatures. The fan could have been eliminated from the TCS design by removing the components from the subsystem enclosures and strategically locating the components by directly mounting the hottest components to the coldest locations of the bus. For example, all of the enclosures were mounted to the electronics shelf for the bus. The resulting temperature on the shelf was 286 K; whereas, the temperature at the top of the bus was 265 K. By mounting

the hottest components to the top of the bus, the thermal balance of the bus would be improved, and the temperatures would be within the operational limits without the need for convection cooling. However, the tradeoff is an increase in the survival heater power from 150 W to 200 W, which leads to the second function of the fan.

In addition to increasing the heat transfer rate of the system, the fan functioned as a heat switch. For the hot case, the fan provided additional cooling to increase the heat transfer rate of the subsystem. During the cold case, the fan is switched off, and heat is primarily transferred by conduction through the enclosure. The result is a significant reduction in survival heater power. From the energy balance, the survival heater power was estimated to be 230 W. For the original worst hot case and worst cold case conditions, the actual survival heater power was 150 W. The significant reduction from the estimated to the actual heater power is a product of enclosing the subsystems in separate electronics boxes and using the fan to adjust the heat transfer rate. Even with the significant reduction from the estimated heater power to the actual heater power, the power level was too high. As a result, more realistic hot and cold case scenarios were used. The first was the hot case orbit with a β angle of 90° ; the second was the cold case orbit with a β angle of 0° . For both of these orbits, their respective hot and cold cases were evaluated. As a result, the heater powers were reduced to more modest levels.

Ideally, a robust TCS would be applicable to both the worst hot case and worst cold case conditions with only minimal heater power. In the final design, the walls of the enclosures and the base plates were assumed to be in perfect contact, and an edge conductivity of 185 W/m-K was used. This design provided three heat transfer paths during the hot case and two paths during the cold case. For the hot case, heat was transferred by convection, conduction through the enclosure, and radiation from the enclosure to the interior of the satellite. For the cold case, convection was eliminated, but the conduction and radiation paths were still present. The result was a large heater power need for the cold case. The design used the fan to help reduce the heater power, but it did not take full advantage of the forced convection thermal switch concept. It is the focus of future work to investigate an alternative design where the heat switch ability of the fan is maximized, and the conduction and radiation paths are minimized.

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REFERENCES

1. Steinberg, Dave S., Cooling Techniques for Electronic Equipment, 2nd Ed., John Wiley & Sons, Inc., New York, 1991.
2. Lyke, Jim, et al., "Space Plug-and-Play Avionics," AIAA 3rd Responsive Space Conference, Paper No. RS3-2005-5001, Los Angeles, CA, 25 – 28 April 2005.
3. Williams, Andrew., Robust Satellite Thermal Control Using Forced Air Convection Thermal Switches for Operationally Responsive Space Missions, Master's Thesis, University of Colorado, Department of Aerospace Engineering Sciences, Boulder, CO, 2005.
4. Griffin, Michael D. and James R. French, Space Vehicle Design, AIAA Education Series, Washington D. C., 1991, pg 392-394.
5. Wertz, James R., and Wiley J. Larson, Space Mission Analysis and Design, 3rd Ed. Microcosm Press, El Segundo, CA, 1999, pg 433.
6. k-Technology "k-core" retrieved from <http://www.k-technology.com/kcore.html> on 21 June 2005.
7. Gilmore, David G., Spacecraft Thermal Control Handbook Volume I: Fundamental Technologies, 2nd Ed, The Aerospace Press, El Segundo, CA, 2002, pg 271-272.
8. Sofasco, "DC Axial Fans," retrieved from <http://www.sofasco.com/dc.htm> on 15 June 2005.